

## **FLOATING CABLE STOP FOR A CABLE ACTUATED BICYCLE COMPONENT**

### 5      RELATED APPLICATIONS

This application claims priority from United States Provisional Patent Application Serial No.60/195,560, filed April 6, 2000, entitled "Mechanical Disc Brake Caliper", the contents of which are incorporated herein in their entirety.

### 10     TECHNICAL FIELD

The present invention is directed toward cable actuated bicycle components, and more particularly toward a bicycle cable feed system and method for routing a flexible cable to a cable actuated bicycle component.

### 15     BACKGROUND ART

Bicycles typically have a number of cable actuated components. These include by way of example, the front and rear derailuers and the brakes. Typically in bicycles a cable extends between these cable actuated components and an actuator applying select amounts of tension to the cable. In the case of derailuers, the actuator is a shifter and in the case of brakes, the  
20     actuator is a brake lever. In order for these cable actuated components to function properly, tension must be applied to the cable. If the cable could be routed between the components in a straight line, this would be a trivial matter. However, because of physical limitations on the location of the cable actuated components and the need to make the actuators readily accessible to users, the cables must be routed in a tortuous path between the actuator and the  
25     cable actuated component.

In part the routing is accomplished by flexible housing which can be bent to accommodate variable curves while maintaining a fixed length. The cable is received in the housing so that cable path can be bent as needed to direct it toward the component without the length of the housing collapsing along its length under actuating forces so as to maintain  
30     tension in the cable. Often the cable can be routed along a straight line parallel to a portion

of the bicycle frame for a portion of its path. In these instances, it is common to provide cable housing stop pairs on the frame which the flexible cable housing axially abuts and the cable extends on a straight line between cable housing stops parallel to the portion of the bicycle frame. While these cable housing stops are effective in minimizing the required use of the flexible cable housing, they must be located and affixed during frame manufacture. Thus, these cable housing stops are not always suitable for routing cable where the location of the actuator or the cable activated component changes due to introduction of new and different components.

One concrete example has been the recent trend of cable actuated disc brakes replacing rim brakes. While most of the cable routing is the same for cable actuated disc brakes and rim brakes, cable actuated disc brakes are located much closer to the wheel hub, requiring between 10 and 14 more inches of cable to extend between the disc brake and the lever. This span is along a straight line but cannot be spanned by cable housing stops due to the absence of frame components along the cable route. Thus, to date, conventional flexible cable housing has been used to span this distance. However, due to the length of the span and the compressive forces applied to the cable housing when the brakes are actuated, the cable housing radially buckles about its axis creating contact between the interior of the cable housing and the cable, which causes friction on the cable that can make the disc brakes more difficult to actuate. While more efficient and powerful cable actuated disc brakes are evolving, to date manufacturers of cable actuated disc brakes have not adequately addressed minimizing friction in the cable system.

Reliance on cable stops mounted to the frame is also a problem because the cable is directly exposed to the environment and, therefore, presents a potential source for grit and other debris to enter the cable housing and thus further exacerbate the friction problems discussed above.

The present invention is directed toward overcoming one or more of the problems discussed above.

## SUMMARY OF THE INVENTION

A bicycle cable guide system for maintaining axial tension in a flexible cable extending between a cable actuated bicycle component and a cable actuator selectively applying tension to the cable includes a first length of flexible housing having a select outer diameter and an inner diameter greater than the diameter of the cable and a straight length of axially and radially rigid tubing having an inner diameter greater than the diameter of the cable. A ferrule joins an end of the first length of flexible housing to a first end of the axially and radially rigid tubing. The cable extends through the first length of flexible housing and the straight length of axially and radially rigid tubing. A second length of flexible housing having substantially the same inner and outer diameter as the first length may have an end joined to the second end of the axially and radially rigid tubing by a second ferrule. Alternatively, the cable actuated bicycle component may include a component cable guide fixed to the component for guiding cable for operative engagement with the component along a guide axis. A second end of the length of axially and radially rigid tubing axially engages the component cable guide along the guide axis with the component guide securing the second end against axial movement toward the component along the guide axis. The cable actuated bicycle component may be a cable actuated disc brake.

A second aspect of the present invention is a method of routing a flexible cable between a cable actuated bicycle component and a cable actuator selectively applying tension to the cable, wherein at least a portion of the flexible cable route is a straight line of a select length. The method includes providing a straight length of axially and radially rigid tubing having a length between ends substantially equal to the select length of the straight line portion of the flexible cable route, feeding the portion of flexible cable corresponding to the straight line portion of the flexible cable route through the axially and radially rigid tubing and fixing the ends of the axially and radially rigid length of tubing against axial movement relative to the cable. The method may further include abutting at least one end of the tubing to an end of a length of flexible housing receiving the cable which is fixed against axial movement relative to the cable. Both ends of the tubing may be abutted by an end of a length of flexible housing receiving the cable which is fixed against axial movement. Alternatively,

one end of the tubing may be abutted to a length of flexible housing receiving the cable which is fixed against axial movement and the other end may be abutted to a component cable guide of the cable actuated component which is fixed against axial movement relative to the cable. The cable actuated bicycle component may be a cable actuated disc brake.

The cable guide system and method of routing flexible cable of the present invention enables the elimination of straight lengths of conventional cable housing without the need of housing stops on a bicycle frame. By eliminating lengths of conventional flexible cable housing which is subject to lengthwise buckling under compression, an unnecessary but significant source of friction on the cable can be eliminated. The axially and radially rigid tubing allows the cable to be self centered within the tube, thereby virtually eliminating significant contact with the interior of the tube and essentially eliminating friction over the length of the tube. By way of contrast, conventional cable housing buckles under the influence of lengthwise compression and the buckling creates a tortuous path for the cable with friction spots along the wall of the housing. The axially and radially rigid tubing further forms a protective seal around the cable that helps prevent the entry of moisture and grit into other portions of the cable routing system, such as conventional cable housing, which can further increase friction on and wear of the cable. The tubing is both lighter and less expensive than standard cable housing, and thus these many advantages can be realized at a cost savings. The use of a cable guide system including the axially and radially rigid tubing has special utility for the use of cable actuated disc brakes which have between 10 and 14 inches or more of cable than rim brakes which traverses a straight path along the radius of a bicycle wheel. Replacing conventional flexible cable housing with the axially and radially rigid tube eliminates an unwarranted source of friction that can make actuation of the cable actuated disc brakes more difficult than necessary.

#### BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a perspective view showing the ball bearing mechanical disc brake caliper of the present invention mounted to a fork of a bicycle in operative engagement with a brake disc;

Fig. 2 is the ball bearing mechanical disc brake caliper of Fig. 1 including an adaptor for mounting to a frame with different mounts;

Fig. 3A is a front elevation view of the ball bearing mechanical disc brake caliper of Fig. 1 including a floating cable stop;

5 Fig. 3B is identical to Fig. 3A except it further includes an alternate embodiment of the floating cable stop;

Fig. 3C is a cross-section of the floating cable stop taken along line 3C-3C of Fig. 3A;

Fig. 4A is an exploded perspective view of the ball bearing mechanical disc brake caliper of Fig. 1;

10 Fig. 4B is an exploded perspective view from a perspective rotated 180° from that of Fig. 4A;

Fig. 4C is a bottom perspective view of a clamp plate in accordance with the present invention;

15 Fig. 5 is a cross-section of the ball bearing mechanical disc brake caliper taken along line 5-5 of Fig. 3A with the brake pads retracted;

Fig. 6 is the same as Fig. 5 only with the brake pads extended using the pad wear compensation apparatus;

Fig. 7 is the same as Fig. 5 only it illustrates the brake pads advanced by the drive mechanism into contact with a disc;

20 Fig. 8 is a cross-section of the ball bearing mechanical disc brake caliper taken along line 8-8 of Fig. 3A;

Fig. 9 is a cross-section of the ball bearing mechanical disc brake caliper taken along line 9-9 of Fig. 8;

25 Fig. 10 is a right side view of the ball bearing mechanical disc brake caliper with the lever arm in an at rest position;

Fig. 11 is a right side elevation view of the ball bearing mechanical disc brake caliper with the lever arm actuated to the braking position;

Fig. 12 is a cross-section of the cable feed taken along line 12-12 of Fig. 10;

Fig. 13 is a front exploded view of the cable feed;

Fig. 14 is an exploded view of the outer indexing knob assembly;

Fig. 15 is an exploded view of the inner indexing knob assembly;

Fig. 16A is a perspective view of the ball bearing mechanical disc brake caliper with a portion of the housing cut away to reveal the pad receiving cavity;

5            Fig. 16B is a sectional view of the ball bearing mechanical disc brake caliper taken along line 16B-16B of Fig. 10;

Fig. 17A-C are alternate embodiments of the backing plates of the brake pad assemblies;

10           Fig. 18 is identical to Fig. 16, only showing the pad assembly installed with the pad assembly recess;

Fig. 19 is a perspective view of the outer knob;

Fig. 20 is a perspective view of the outer knob from a perspective rotated 180° from that of Fig. 19;

Fig. 21 is a perspective view of the inner knob;

15           Fig. 22 is a perspective view of the inner knob taken from a perspective rotated 180° from that of Fig. 21;

Fig. 23 is a front view of the lever arm illustrating the progressive, eccentric shape of the cable guide surface;

20           Fig. 24 is a front view of the lever arm illustrating the constant, concentric shape of the cable guide surface;

Fig. 25 is a perspective view of a ball retainer;

Fig. 26 is a sectional view of a ball retainer taken along line 26-26 of Fig. 25 with a ball engaged therein;

Fig. 27 is a perspective view of an alternate embodiment of a ball retainer;

25           Fig. 28 is a sectional view taken along line 28-28 of Fig. 27 with a ball engaged by the retainer; and

Fig. 29 is a plan view of an alternate embodiment of ramped grooves in a fixed cam.

## DETAILED DESCRIPTION OF THE INVENTION

5 A ball bearing mechanical disc brake caliper 10 in accordance with the present invention is shown in Fig. 1 mounted to a frame or, more particularly, a front fork 12 of a bicycle in operative engagement with a disc 14. As shown in Figs. 1-3, the caliper 10 is mounted to a front fork 12 for use with a front wheel. For use with the rear wheel, the caliper is typically mounted to the seat stay, chain stay, drop out plate, after market adapter or the like. The disc 14 in turn is rigidly mounted to the hub of a wheel assembly by the bolts 16. For the sake of clarity, the bicycle wheel and hub are not shown.

10 The ball bearing mechanical disc brake caliper consists of a caliper housing 18 having a pair of mounting feet 20, 22 extending therefrom for attachment to a corresponding pair of internally threaded attachment bosses 24, 26 which extend from the front fork 12. A pair of mounting bolts 28 secure the mounting feet 20, 22 to the attachment bosses 24, 26. The mounting feet preferably include elongate slots 27 (see Fig. 5) receiving the mounting bolts 28 and complimentary pairs of concave/convex washers 30 to provide for adjustable  
15 attachment of the ball bearing mechanical disc brake caliper to a bicycle frame. Such an attachment structure is described in detail in applicant Wayne Lumpkin's, co-pending Patent Application Serial No. 09/383,121, the disclosure of which is hereby incorporated in its entirety herein.

20 As seen in Fig. 1, a lever arm 32 is pivotably attached at a first end 34 to the caliper housing 18. A second end of the lever arm 36 has a cable clamp 38 which secures an end of the cable 40. The cable 40 is directed through a cable feed 42 attached to the caliper housing 18 with a cable housing 44 abutting the cable feed 42. While the operation of the ball bearing mechanical disc brake will be described in considerably greater detail below, it is useful at the outset to understand that the ball bearing mechanical disc brake caliper is actuated by tension  
25 being applied to an opposite end of the cable 40 by a cable actuator such as a conventional cable brake lever (not shown) and this tension causes the lever arm 32 to pivot about pivot axis 46 in the direction of arrow 48 so that the second end of the lever arm 36 is drawn toward the cable guide 42 to advance a brake pad into contact with the disc 14 by a rotary to linear linkage between the first end 34 of the lever arm 32 and the brake pad.

Fig. 2 shows the ball bearing mechanical disc brake caliper 10 mounted to a front fork 12' having internally threaded attachment bosses 24' and 26' with an axis parallel to the axis of rotation of the disc 14. The ball bearing mechanical disc brake caliper 10' is in all manner identical to the ball bearing mechanical disc brake 10 described above with regard to Fig. 1.

5 For simplicity, all unnecessary corresponding reference numbers have been omitted. An adapter bracket 60 is fastened by a pair of bolts 62 to the attachment bosses 24', 26' and includes a pair of internally threaded receptor bores 64 that enable the caliper housing 18 to be attached to the front fork 12' in an identical position relative to the disc 14 described above with respect to Fig. 1. Thus, the adapter bracket provides an equivalent mounting surface to  
10 that provided by the attachment bases 24, 26, as shown in Fig. 1.

Fig. 3A is a front elevation view of the ball bearing mechanical disc brake caliper 10 mounted to a bicycle frame 12 as illustrated in Fig. 1. Fig. 3 differs from Fig. 1 by the inclusion of the floating cable stop 70, which will be described in greater detail below.

The ball bearing mechanical disc brake caliper 10 is shown in an exploded perspective view in Fig. 4A. Fig 4B is identical to Fig. 4A, only the perspective is rotated 180°. First and second brake pad assemblies 72, 74 consist of mirror image backing plates 76, 78 each having a trailing surface 80 including a post receiving receptacle 81 and a leading surface 82 to which a brake pad 84 is permanently adhered. When the ball bearing mechanical disc brake caliper is operatively associated with a disc 14, the disc 14 resides between the pads 84 of the first and second brake pad assembly 72, 74 which are held in place in part by a pad retention clip  
15 20 85 in a manner which will be described in greater detail below.

As oriented in Fig. 4A, the second brake pad assembly 74 is also known as the back or inboard brake pad assembly. A pad wear compensator 73 for the inboard pad assembly includes inboard pressure foot 86, which as discussed below, functions as an indicator. The  
25 inboard brake pad assembly 74 is attached to the inboard pressure foot 86 by means of a washer-shaped magnet 88 which is adhered to a cooperatively shaped receptacle 90 in the leading surface 92 of the inboard pressure foot 86. An axial post 94 extends through the hole in the washer-shaped magnet 88 and protrudes beyond the leading surface 92 to engage the post receiving receptacle 81 in the trailing surface of the backing plate. A trailing portion or



indicator dog 96 having a rectangular cross-section extends from a trailing surface of the inboard pressure foot 86 along the same axis of the axial post 94. The edge of the inboard pressure foot 86 is threaded as indicated at 100 between the leading surface 92 and the trailing surface 98. The threads 100 are sized to threadably engage complimentary threads 102 in the inner diameter of an inside cylinder 104 of the caliper housing 18 (see Figs. 4B and 5). This threaded engagement allows for linear advancement of the pressure foot as it is rotated. An inboard pad advancement adjustment knob 106 has knurled edge 108, an axial orifice or hole 110. The axle hole is configured snugly, axially, slidably receive the indicator dog 96 of the inboard pressure foot 86 but to prevent rotation between the pressure foot and the adjustment knob. A plurality of axially inward extending legs 112 having radially outwardly extending barbs 114 at their distal ends. An inside indexing spring clip 116 has a plurality of radially extending legs 118 sized to be received between the axially inwardly extending legs 112 of the inner knob 106. The inside indexing spring clip 116 further includes axially inwardly extending bars 122 having radially outward extending detents 123 at their distal ends. As best seen in Fig. 5, the dog 96 extends through the hole 126 in the inside indexing spring clip 116 and into the axial hole 110 in the inner knob 106. The barbs 114 engage an inner edge of an inward extending annular flange 125 to lock the inner knob 106 against axial movement. The detents 123 in turn engage equally circumferentially spaced indexing knurls 127 in the inner surface of the flange 125. As will be described below, the complimentary detents and indexing knurls provide a tactile indication of pad adjustment as the inboard knob 106 is rotated.

With continued reference to Figs. 4A, 4B and 5, the caliper housing 18 also includes an outboard cylinder 128 which is coaxial with the inboard cylinder 104. The bulk of the remaining components of the ball bearing mechanical disc brake caliper 10 reside within the outboard cylinder 128. The outboard cylinder 128 has an annular groove 130 (see Fig. 4B) in its inner diameter sized to receive the hoop-shaped polymer dust seal 132. Outboard pressure foot 134 has an identical leading surface to the leading surface 92 of the inboard pressure foot 86 and identical reference numbers are used in Fig. 4B. Washer-shaped magnet 88', which is identical to washer-shaped magnet 88, is adhered within the cooperative shaped

receptacle 90 of the leading surface 92 of the outer pressure foot 134. The outside or first brake pad assembly 72 is attached by the washer-shaped magnet 88' to the leading surface of the outer pressure foot 134. The trailing surface 136 has an axially extending post 138 having an annular groove 140 in its sidewall near the distal end. In the distal end is an axial cup 142. Split ring 144 is sized to be received in the annular groove 140. Ball bearing 146 is sized to be received in and to extend axially from the axial cup 142.

An indicator foot screw 148 has a head 149 with a leading surface 150 which abuts the ball bearing 146. Behind the head 149 is a shaft 152 which is threaded adjacent to the head 149 as indicated at 154. The trailing end of the shaft 152 has a pair of flats 156 (one shown in Fig. 4A) on opposite sides. The indicator foot screw 148 is an integral part of a pad wear compensator 153 for the outboard brake pad assembly.

Drive cam 158 has an enlarged diameter base 160 having a plurality of equally spaced curved, ramped grooves 162 in its trailing surface. The preferred embodiment has three ramped grooves 162 spaced at 120°. A cylindrical shaft 164 extends rearward of the enlarged diameter base 160 and has an axial bore 166 which extends axially through the drive cam 158. As best viewed in Fig. 5, the axial bore includes a threaded inner diameter portion 168 which threadably engages the threaded portion 154 of the foot screw 148 with the shaft 152 extending rearwardly from the axial bore 166. Further referring to Fig. 5, an inwardly extending flange 170 acts as a stop against a rearward portion of the head 149. The distal end of the outside cylindrical shaft 164 is threaded at 172 and adjacent the treaded portion 172 is a hex portion 174. One of three ball bearings 176 resides in each ramped groove 162. The outer diameter of the enlarged diameter base 160 is sized to fit snugly within the inner diameter of the outside cylinder 128 and have a sealing relationship with the dust seal 132 as best seen in Figs. 5 and 7.

Fixed cam 178 has a generally cylindrical body 180 with a constant inner diameter orifice 182. An intermediate step 184 has a spring tension limiting boss 186 which extends axially onto the cylindrical body 180. A leading step 188 has an outer diameter greater than that of the intermediate step 184 and an enlarged outside diameter annular flange 190 rises from the leading step 188 adjacent the intermediate step 184. A locking boss 192 extends

toward the leading surface 193 collinearly with the spring tension limiting boss 186 at a height matching that of the enlarged outer diameter annular flange 190. The locking boss 192 is sized to key into a receiving slot 194 in the inner diameter of the outside cylinder 128 to lock the fixed cam 178 against axial rotation (see Fig. 4A). In addition, the leading surface of the enlarged outer diameter annular flange 190 abuts a step 196 in the inner diameter of the outside cylinder 128 to halt axial insertion of the fixed cam 178 into the outside cylinder 128 from the opened end as viewed in Fig. 4A. The engaged relationship can best be seen in Fig. 5. The leading surface 193 of the fixed cam 178 is best viewed in Fig. 4B. The leading surface has a plurality of equally circumferentially spaced ramped grooves corresponding to the ramped grooves of the drive cam 158. Fig. 4B shows three ramped grooves 200 spaced at 120° which correspond to the ramped grooves 162 of the drive cam 158, only with the ramps extending circumferentially in opposite directions when aligned as shown in Figs. 4A, 4B and 5-7. A ball bearing 176 resides between each ramped groove pair 162, 200 as best viewed in Figs. 5-7. Referring to Fig. 5, with balls residing in the grooves 162, 200, the grooves and ball bearings act as an angular contact bearing which is able to accommodate axial loads on the drive cam exerted by the lever arm 32. In addition, the ramped grooves self-center the drive cam shaft 164 within the inner diameter 182 of the fixed cam 178 with the drive cam under an axial load. This feature eliminates the need for an optional split bushing (not shown) being press fit in the inner diameter 182 of the fixed cam 178. It further eliminates friction between the drive cam shaft 164 and fixed cam 178. It further reduces needs for tight tolerances between the drive cam shaft 164 and fixed cam 178, thus eliminating the need for costly centerless grinding of the drive cam shaft and reaming of the fixed cam bore 182. These combined advantages significantly improve performance and minimize parts cost and assembly complexity and attendant cost.

When the fixed cam is seated within the outside cylinder 128 as described above and as viewed in Fig. 5, it is locked against axial movement by locking ring 204 which has a threaded outer diameter 206 and evenly spaced engagement slots 208 in the inner diameter 210. The inner diameter is sized to snugly receive the intermediate step 184 of the fixed cam 178 and the engagement slots 208 allow for engagement by a special turning tool (not shown)

so that the threaded outer diameter 206 can be brought into threaded engagement with corresponding threads 212 in the inner diameter of the outside cylinder 128.

A generally washer-shaped spring tension biasing plate 220 has an inner diameter which snugly axially receives the intermediate step 184 of the fixed cam 178 and includes a spring tension limiting slot 222 which receives the spring tension limiting boss 186. A cut in the outer diameter of the spring tension biasing plate forms a stop surface 224. Near the stop surface 224 is a hole 226. Return spring 228 has a pair of axially extending ends 230, 232. The inner diameter of the return spring 228 is large enough to axially receive the fixed cam 178 and the shaft 164 of the drive cam 158 as best viewed in Fig. 5. The axially extending end 230 is received in the hole 226 of the spring tension biasing plate 220 (see Fig. 5). A dust seal 234 defines an annular cover 236 for the return spring 228 as seen in Figs. 4B and Figs. 5-7. The inner diameter of the trailing orifice 238 is sized to receive and have a sealing relationship with the outer diameter of a leading flange 240 of the lever arm 32. A hole 242 in the trailing surface of the cover 236 receives the axially extending rod 232. The axially extending rod 232 in turn is received in the hole 244 near the first end 34 of the lever arm 32.

A hex orifice 246 near the first end 34 of the lever 32 axially receives the hex portion 174 of the cylindrical shaft 164 of the drive cam 158 with a hex inner diameter washer 248 therebetween to radially fix the lever arm 32 to the drive cam 158. Washer 252 abuts the trailing surface 254 and is sandwiched by a larger outer diameter washer 256. The larger outer diameter washer 256 has a number of equally circumferentially spaced indexing knurls 258 in its outer diameter. The washers 252, 256 and the lever arm 32 are axially secured to the cylindrical shaft 164 of the drive cam 158 by nut 260 which threadably engages the threaded portion 172 of the cylindrical shaft 164.

An outboard knob 264 has a knurled edge 266 and an orifice or axial hole 268 sized and dimensioned to snugly receive the flats 156 of the trailing end of the foot screw 148 therein, as illustrated in Fig. 5. Referring to Fig. 4B, a plurality of axially inwardly extending legs 270 are equally circumferentially spaced in an inside surface of the outer knob 264. At the distal end of each axially inwardly extending leg 270 is an inwardly protruding barb 272.

An outside indexing spring clip 274 has a plurality of axially extending bars 276 each having an inwardly extending detent 278 near its distal end. The axially extending bars 276 are sized to snugly fit between the axially inwardly extending legs 270 of the outboard knob (see Fig. 4A). With the outside indexing spring clip axially engaged with the outer knob 264 in the orientation illustrated in Fig. 4A, the outer knob 264 is axially advanced over the nut 260 and the inwardly protruding barbs 272 lockingly engage the outer diameter edge of the large outer diameter washer 256 to lock the outer knob 264 against axial movement. When attached in this manner, the inwardly extending detents 278 of the outside indexing spring clip engage the indexing knurls 258 of the larger outer diameter washer 256. This can be best seen in detail with reference to Figs. 14 and 5. As will be described further below, the complimentary detents and indexing knurls provide a tactile indication of pad advancement as the outboard knob 264 is rotated.

With reference to Figs. 4A, 12 and 13, the cable feed 42 consists of a mount 284 which is preferably integrally cast with the housing 18. The mount 284 includes an orifice 286 centered along a guide axis 288. A cylindrical housing stop ferrule 290 has a cylindrical main body 292 having an outer diameter dimensioned to fit freely yet snugly within the orifice 286. A minor boot retention barb 294 extends axially from a leading end of the housing stop ferrule 290. A major boot retention barb 296 extends axially from a trailing end of the housing stop ferrule 290. An annular retention flange 298 extends radially from the main body 292 adjacent to the major boot retention barb 296 and forms a stop which halts axial insertion of the housing stop ferrule 290 into the orifice 286, as best seen in Fig. 12. Further referring to Fig. 12, the inside of the housing stop ferrule 290 has a trailing portion having an inner diameter slightly larger than that of a standard cable housing to axially receive the cable housing 44 therein. An annular flange 302 extends inwardly to define a cable guide orifice 304. The inner diameter of the minor boot retention barb 306 is of a size between that of the trailing inner diameter 300 and the cable guide orifice 304.

A hollow minor retention boot 310 is molded of an elastimeric material and at its trailing edge has an inwardly extending annular flange 312 configured to lockingly engage with the minor boot retention barb 294 of the housing stop ferrule 290. With the housing stop

ferrule 290 inserted in the orifice 286 as illustrated in Fig. 12 and the minor retention boot mounted with the inwardly extending annular flange 312 engaging the minor boot retention barb 294, the housing stop ferrule is secured against removal from the orifice 286. The minor retention boot has a leading nipple 314 having a leading hole 316 with an inner diameter slightly less than the outer diameter of the standard bicycle brake cable 40. In this manner, the leading nipple forms a wipe seal with the brake cable 40 as seen in Fig. 12.

A hollow major retention boot 320 molded of an elastomeric material has an inwardly extending annular flange 322 sized to lockingly engage with the major boot retention barb 296 on the trailing end of the housing stop ferrule 290 as best viewed Fig. 12. The trailing end 324 has a tapered inner diameter, which at the extreme trailing end is slightly smaller than the outer diameter of the standard cable housing to form a sealing relationship therewith.

With the lever arm 32 pivotably attached to the housing as illustrated in Figs. 1-3B, 10 and 11, a curved horn 330 defining an axially flat, circumferentially curved cable guide surface 332 extends from a trailing end of the second end 36 of the lever 32. The curved horn 330 curves about the axis of rotation 46 of the lever arm 32. In the preferred embodiment, the curved horn is eccentric about the axis as illustrated schematically in Fig. 23 to provide for progressive increase in power as the lever is actuated by a cable 40. Alternatively, the curved horn can be concentric as shown in Fig. 24 or eccentric and regressive, which though not illustrated, would require the curved horn to have an increasing radius as it extends toward its free end, essentially the opposite of the progressive horn illustrated in Fig. 23.

The cable clamp 38 consists of a screw 334 having a threaded shaft 336 sized to threadably engage an internally threaded bore in the lever arm 32 having an axis normal to the axis of rotation 46. In the preferred embodiment, a clamp plate 338 is secured between the head of the screw 334 and the second end 36 of the lever arm 32. The clamp plate has a tab 340 which is received in a notch 342 defined in the distal end of the lever arm 32 to fix the clamp plate against rotation. A groove 344 is formed in the underside of the clamp plate adjacent to the notch 342 to receive the cable 40 and has a number of protrusions 345 extending therein to improve the grip of the cable, as illustrated in Fig. 4B.

The curved horn 330 is configured so that with the ball bearing mechanical disc brake caliper installed on a bike frame as illustrated in Figs. 1-3B, the guide axis 288 is essentially tangent to the free end of the curved horn 330. Essentially tangent means a cable 40 does not have a significant bend when it contacts the cable guide surface 332, but instead has a very gradual transition to the cable guide surface 332 as viewed in Fig. 3. When tension is applied to the cable 40 by a tension actuator such as a conventional bicycle brake lever, the lever arm 32 is drawn toward the cable feed 42. Because of the circumferentially curved cable guide surface 332, the fixed cable clamp and the fixed cable feed 42, no sharp bends are introduced to the cable 40 which might fatigue the cable and lead to premature failure of the cable, which could have disastrous results for a user.

In the embodiment illustrated in Fig. 1, the conventional cable housing extends from the trailing end of the major retention boot 320. An improvement to this conventional brake setup is to provide a floating cable stop 70 mating with the trailing inner diameter 300 of the housing stop ferrule 290 as illustrated in Fig. 3A as part of a bicycle cable guide system. The floating cable stop 70 consists of a axially and radially rigid tube 348 made of a suitable material such as a metal like aluminum or stainless steel or an exceptionally rigid thermoplastic. As used herein, axially and radially rigid means the tube 348 has sufficient rigidity that it will not radially buckle about its lengthwise axis upon application of tension within the normal range of operating tensions applied to the cable 40 which runs within the tube 348. In the preferred embodiment, the tube 348 has a standard cylindrical cross-section (see Fig. 3C), although other cross-sections may be useful or desired. The outer diameter is preferably essentially the same as that of a standard cable housing 44 so that it can fit into a trailing end of the housing stop ferrule 290 in the same manner as the housing 44, as illustrated in Fig. 12. This forms an axially fixed connector for the tube 348. A connector ferrule 350 connects the tube 348 to a conventional cable housing 44. The conventional cable housing 44 is configured to bend along its length but to maintain a fixed length. This combination forms another axially fixed connector for the tube 348. The conventional cable housing allows the cable to be bent as may be required to attach the cable to a brake lever. A significant advantage of the floating cable stop 70 is that when it replaces conventional

cable housings, it provides a straight path for the cable inside with minimal or no contact with the inner diameter of the tube. Over all but the shortest of lengths, the flexible cable housing will radially buckle about the lengthwise axis under application of even minor tension to the cable within and the resultant compression to the cable housing. Elimination of this buckling further reduces contact of the cable with the inner diameter of the tube and serves to further minimize friction on the cable. The floating cable stop can be deployed wherever there is a straight length of cable, independent of fixed housing stops on the bicycle frame. It also provides a protective barrier for the cable, much like conventional flexible cable housing, but at a lesser weight.

In a preferred embodiment illustrated in Fig. 3B, a small length of conventional flexible housing 352 is disposed between the tube 348 and the housing stop ferrule 290 and is joined to the tube 348 by connector ferrule 354 to form an axially fixed connector. The transition housing 352 is advantageous because it will laterally flex or bend in the event of a lateral blow to the tube 348 and thereby minimize the risk of bending of the tube 348 which would detract somewhat from its performance and could even result in undesired radial buckling of the tube 348. Preferably, the transition housing 352 is of a length that will not radially buckle under application of operating tensions to the cable 40 but will still provide sufficient radial flexibility to protect the tube 348. Alternatively, if required, the transition housing 352 could be long enough to bend the cable as required to properly direct the cable to the cable feed. Or, an apparatus such as the ROLLAMAJIG, manufactured to Avid, L.L.C., of Englewood, Colorado, U.S. Patent No. 5,624,334, the disclosure of which is hereby incorporated by reference, could be substituted for the transition housing to minimize friction where a bend is required to direct the cable.

It should be apparent to those skilled in the art that floating cable stop 70 could be deployed on any cable actuated bicycle component, including cantilevered brakes, caliper brakes, side pull cantilever brakes and derailuers.

The first and second brake pad assemblies 72, 74 are made to be removable from the caliper housing when a rotor is not operatively associated with the caliper housing between the brake pad assemblies. Referring to Fig. 16A, a retention structure for the first and second



brake pad assembly 72, 74 is illustrated. The caliper housing has a cavity 360 configured to receive the disc or rotor 14. The cavity 360 has a mouth 362 at a leading end and includes a pair of opposing recesses 364 (one shown in Fig. 16A). The recesses 364 are configured to nest the backing plates 76, 78 of the brake pad assemblies 72, 74 on opposite sides of the disc so that the friction pads 84 can be brought into and out of engagement with the disc by an actuating or drive apparatus along an advancement axis 366 in a manner that will be described in greater detail below. At a leading end 368 of pad assembly 72 is a retention tab 370 formed from a pair of extending posts 372, 374 having oppositely extending protrusions 376. Referring to Fig. 16B, within the cavity 360 opposite the mouth 362 is a retention clip cavity 378 opening into the cavity 360. Engagement flanges 380 extend from opposite sidewalls of the retention clip cavity. Pad retention clip 85 is shown in Fig. 16A installed within the retention clip cavity 378. The pad retention clip 85 has a base 382 with a pair of extending sidewalls or legs 384, 386 with a retention detent 388 near the far end of each leg protruding inwardly. Near the base 282 a plurality of retention barbs 390 extend laterally from the sidewalls or legs 384, 386. As illustrated in Fig. 16B, these retention barbs 390 are configured to snap fit with the engagement flanges 380 to lock the pad retention clip 85 within the retention clip cavity 378.

Referring back to Fig. 16A, the pad assembly 72 is installed by grasping the handle 392 and advancing the leading edge 368 into the mouth 362 along the engagement axis 394 and aligning the retention tab 370 with the pad retention clip 85 and further advancing the pad assembly so that the protrusions 370 mate within the retention detents 388. The pad can then be slid into the recess 364 along the advancement axis 366 to seat the pad assembly within the recess 364, as viewed in Fig. 18. When seated in this manner, the walls of the recess 364 secure the pad assembly against movement transverse the advancement axis 366 as a rotating disc is engaged. As best viewed in Fig. 5, it should be appreciated that the axial post 94 of the respective inboard or outside pressure foot 86, 134 is received within the receptacle 81 and the trailing surface 80 of the backing plates to thereby prevent withdrawal of the pad assembly from the mouth 362 of the cavity 360 with the brake pad seated as illustrated in Fig. 18. This connection is also the primary support against withdrawal along the engagement axis as the

pad assembly is advanced and withdrawn by the actuation mechanism. The magnet 88 or 88' holds the backing plate in abutment with the respective pressure foot 86, 134 to maintain engagement between the axial post 94 and the receptacle 81. As the brake pads are advanced along the advancement axis, the cooperating engagement flanges 380 of the pad retention clip and the protrusions 376 of the pad retention tab define a rail facilitating movement forward and backward along the advancement axis. The pad clips can be easily removed from the orifice simply by manually advancing them inward along the advancement axis to bring the receptacle 81 out of engagement with the axial post 94 whereupon the engagement flanges 380 can be snapped out of engagement with the protrusions 376. As shown in Figs. 16A and 16B, the handle 392 has straight edges. To facilitate gripping, the handle may be modified as shown in Figs. 17A-C. In Fig. 17A, the handle has a distal enlargement 395. In Fig. 17B, the handle has grooves 396. In Fig. 17C, the handle has knurls or bumps 397. Other grip enhancing structures will also be apparent to those skilled in the art.

The operation of the ball bearing mechanical disc brake caliper 10 drive mechanism is best understood with reference to Figs. 1, 5, 6, and 7. Upon actuation of the lever arm 32 by tension applied to the cable 40, the lever arm rotates about the pivot axis 46 in the direction of arrow 48. This in turn causes rotation of the drive cam 158 about this same axis. As the drive cam 158 is rotated, the ball bearings 176 cause the drive cam to advance within the outside cylinder 128 which in turn advances the foot screw 148 which is threadably engaged with the drive cam. The leading surface 150 of the foot screw 148 in turn advances the ball bearing 146 and the outside pressure foot 134 to urge the pad 84 of the outside brake pad assembly 72 into contact with the disc. Further advancement will deflect the disc 14 into contact with pad 84 of the outside brake pad assembly 74, as illustrated in Fig. 7. Upon release of the tension in the cable, the lever arm is biased back to its at rest position by the return spring 228 and the pads are retracted out of contact with the disc to reassume the position illustrated in Fig. 5.

Fig. 10 illustrates that with the lever arm 32 in an at rest position, the cable extends between the cable clamp 38 and the cable feed 42 at a slight angle. With the lever arm 32 rotated about the pivot axis in the direction of arrow 48 so as to bring the pads into

engagement with the disc, the lever arm advances axially along the advancement axis with the outer brake pad assembly 72 so that this slight angle is eliminated, as seen in Fig. 11. Thus, it is desirable that the axially flat, circumferentially curved cable guide surface 332 be wide enough in the axial direction to accommodate the axial movement of the lever arm 32. A the pads wear, it may be necessary or desirable to advance the pressure feet within the inboard and outboard cylinders to maintain the original spacing between the pads and the disc. The present invention provides a pad wear compensating apparatus that allows for such advancement (or retraction) by rotary to linear linkages between the knobs 106, 264 and the respective pressure feet 86, 134 and associated pads.

As described above, the pad wear compensator includes inboard pressure foot or inboard indicator 86 which is threadably engaged with the sidewall of the inside cylinder. Rotation of the inner knob 106 in a clockwise direction advances the pressure foot within the cylinder and therefore the pad assembly along the advancement axis as illustrated in Fig. 6. As the pressure foot is advanced, the trailing end or indicator dog 96 received in the axial hole 110 of the inside knob 106 advances, to provide both a visual and tactile indication of the amount the pressure foot has advanced within the inside cylinder. In addition, the radially outwardly extending detents 124 of the inside indexing spring clip 116 engage with equally circumferentially spaced knurls 126 in the inner diameter of the flange 125 to provide a tactile indication of movement of the knob. The knurls 126 and radially outwardly extending detents 124 are spaced so that each engagement between the detents and sockets indicates a uniform linear distance of advancement of the pad toward the disc. For example, in the preferred embodiment, each tactile click equates to 1/16 of a full rotation and 1/16 of a millimeter of pad advancement. The inboard pad assembly is retracted by rotating the inside knob counter-clockwise.

The outboard pad wear compensation apparatus 153 relies on a similar rotary to linear linkage as the inboard pad compensator 73, but it is slightly more complicated. Rotation of the outside knob 264 in a clockwise direction in turn causes rotation of the indicator foot screw 148 in a clockwise direction. This rotation threadably advances the indicator foot screw 148 relative to the drive cam 158 which in turn advances the ball bearing 146, the outside

pressure foot 134 and the corresponding first brake pad assembly 72. The outside pressure foot in its advanced position is illustrated in Fig. 6. It should be noted that the split washer 141 received in the annular groove 140 causes friction between the outside pressure foot and the fixed cam to prevent the outside pressure foot from simply sliding out of the outside cylinder. As with the inside knob, the outside knob also provides a tactile indication of rotation corresponding to a select linear advancement. This is provided by the inwardly extending detents 278, which engage with the indexing knurls 258 of the larger outer diameter washer 256. In addition, as described above, advancement of the indicator foot screw 148 and therefore the outside pressure foot 134 can be monitored visually and by feel by noting how far the trailing end 156 of the indicator foot screw 148 advances relative to the outer surface of the outer knob 264 within the axial hole 268. To retract the pad, the outside knob is rotated counter-clockwise to retract the indicator foot screw 148 and the drive mechanism is actuated to squeeze the disc, which in turn retracts the outside pad assembly 72 and the outside pressure foot 134 by forcing them into abutment with the retracted foot screw 148.

The pad wear compensating apparatus not only allows for convenient advancement of the brake pad assemblies as the brake pads wear, but the structure also provides a quick and convenient way to properly align the caliper housing 18 relative to a disc 14. This can be done by loosening the mounting bolts 28 and then advancing the pad assemblies into contact with the disc using the inboard and outboard pad wear compensators 73, 153. With the disc squeezed between the pads, the mounting structure including the slotted mounting feet 20, 22 and the concave and convex washers 30 enables precise alignment of the caliper housing to maintain the leading pad surfaces parallel to the disc. Tightening the mounting bolts 28, 30 then secures the precise alignment. For example, because the inner pad assembly is stationary, it is generally preferred to provide a very small clearance between the inner pad and the disc and a greater clearance between the moveable outer pad and the disc. This set up can be achieved by starting with the pads fully withdrawn along the advancement axis into the cavity 360 as shown in Fig. 5 and then advancing the inner pad assembly using the inner knob a short distance while advancing the pad associated with the outer knob a greater

distance into contact with the disc. The mounting bolts are then tightened and the knobs are turned to retract the pad assemblies to provide a desired operative gap with the disc.

While this greatly simplifies the process of properly aligning the caliper housing and brake pads during initial set up, the pad advancement structure in combination with the caliper housing mounting system also provide for simplified field repair. For example, if a user crashes and one of the attachment bosses is bent, the user can detach the mounting bolts 28, bend the bent attachment boss back in position as well as possible by eye-balling it and then reposition the caliper housing with the brake pads properly aligned parallel to the disc simply by repeating the procedure described in the preceding paragraph.

In operation, as the brake pads are caused to compress the disc therebetween, a high tensile force is applied to the housing in the vicinity of the inside and outside cylinders. This can put tremendous stress on the housing, and can even cause the housing to split apart. This problem is all the more acute where the housing is cast from a lightweight, relatively low tensile strength metal such as aluminum. To address this problem, the ball bearing mechanical disc brake housing has a pair of threaded bores 400, 402, which extend the width of the housing on opposite sides of the pivot axis 46. A steel screw 404 threadably engages each threaded bore 400, 402 and is tightened to pre-stress or pretension the caliper housing. The screws are preferably tightened to apply a compression force of about 1,000-1,400 lbs. This not only prevents cracking and failure of the housing, it virtually eliminates any flexure of the housing that could dissipate braking power or fatigue the housing.

The ball bearing mechanical disc brake caliper 10 also includes a mechanism for adjusting the return force on the lever arm 32 applied by the return spring 228. Referring to Fig. 9, adjustment screw 410 is threadably received in a threaded bore 412 in the housing which breaches the outer cylinder with the axis of the bore 412 aligned with the stop surface 224 of the spring tension biasing plate 220. As the adjustment screw 410 is advanced within the treaded bore 412, the spring tension biasing plate 220 rotates about the cylindrical body 180 of the fixed cam 178 to increase the tension on the spring. Turning the adjustment screw 410 to retract it from the bore causes rotation of the spring tension biasing plate 220 which decreases the tension on the spring 228. As seen in Fig. 9, the spring tension limiting slot 222

cooperates with the spring tension limiting boss 186 of the fixed cam 178 to limit rotation of the spring tension biasing plate 220 and therefore the range of return force applied to the lever arm 32.

It may be useful or desirable to provide a ball spacer between the drive cam 158 and the fixed cam 178 to maintain the ball bearings 176 equally spaced within the elongated ramped grooves 162, 200. If such a ball spacer is to be used, one embodiment of a design for such a ball spacer is illustrated in Figs. 25 and 26. The ball spacer 420 could be made of a simple sheet metal stamping consisting of a ring body 422 with inwardly extending radial leg pairs 424 spaced to correspond to the desired spacing of the ball bearings. The radial legs 424 can be curled as illustrated in Fig. 25 to define a ball receiving socket 426. The legs 424 of each pair are circumferentially spaced so that a ball bearing 176 can be snap fit therebetween as illustrated in Fig. 26. Figures 27 and 28 depict another embodiment of a ball spacer molded of plastic. Notches 427 in a ring 428 are sized to snap fit with the ball bearings 176. The ring is thick enough and the insides of the notch are slightly concave (see 429 in Fig. 27) to secure the ball bearing about an axis as illustrated in Fig. 28. Either ball spacer embodiment secures the ball bearings 176 about an axis and thus ensures that the ball bearings 176 will maintain an equal radial spacing and further ensures that the ball bearings will be the same distance between the face of the drive cam and the fixed cam.

The ramped groove structure of the fixed cam and drive cam illustrated in Figs. 4A and 4B is useful for most applications, but it limits the amount the lever arm can be rotated to, at most, slightly under 120°. An alternate ramp structure is depicted in Fig. 29. As illustrated in Fig. 29, the ramps 430 spiral inward as they ramp upward toward the leading surface 432. With such corresponding structures provided in the leading surface of the fixed cam and the drive cam, the ramped grooves 430 can be much greater in length and have a much more gradual incline. This will enable the associated lever arm 32 to rotate much greater than 120° and for the inboard brake pad to be advanced linearly at a slower rate as the lever arm 32 is pivoted.

The outer knob is shown in a perspective view in Fig. 19. The outer knob has an elongate slot 440 corresponding to each axially inwardly extending leg 270. Referring to Fig.

20, each elongate slot 440 overlies a corresponding barb 272. The holes 440 are formed during molding of the outer knob 264 by a mandrel which occupies the space that defines the hole 440, with the distal end of the mandrel contributing to the forming of the undercut of the barb. In this manner, the undercuts are introduced to the knob while still enabling the knob to be injection molded in a single step. Referring to Figs. 21 and 22, the inner knob 106 likewise has elongate slots 442 corresponding to each inward axially extending leg 112. As with the outer knob described above, the slots overly the barbs 114 and enable formation of the undercut on the barbs by means of mandrels as described above with regard to the inner knob.

10